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No. 76

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APPLIED SCIENCES

PROPERTIES OF HYDROSTATIC BEARINGS, OPTIMUM DESIGN ANALYZED

Nanjing NANJING GONGXUEYUAN XUEBAO [JOURNAL OF NANJING INSTITUTE OF TECHNOLOGY] in Chinese No 1, 1980 pp 97-116

[Article by Xu Shangxian [6079 1424 6343]: "Analysis of the Properties of Hydrostatic Bearings and Optimum Design." This article was received on 22 December 1979]

[Text] Abstract

This article discusses the hydrostatic centripetal slide bearings of the capillary and orifice throttled multiple oil recess and circumferential oil groove. The properties of any number of recesses are analyzed based on the equation of continuity of flow to derive the generalized formulas for computer computation of the pressure in the recess, load capacity and stiffness, amount of flow and power loss. This article also explains the effects of some nondimensional design parameters upon the properties of the bearings.

Because of higher and higher speeds of rotation of the shaft, heating and the rise in temperature of the bearings is a critical problem. Therefore, for a given geometrical dimension of the recess, the computer has been used to calculate the optimum design under the constraints of maximum load capacity and stiffness using power loss as the target function. The penalty function is used to reduce the problem of constrained optimization to a series of problems of unconstrained optimization and Powell's method is used to obtain the optimum solution. A detailed flow chart and FORTRAN program have also been written.

I. Foreword

Hydrostatic bearings rely on the oil pump to provide high pressure oil to the space between the shaft and the bearings forcefully forming an oil film. This in turn provides liquid lubrication at low speeds or even when the shaft is stationary and not rotating. Therefore they possess superior qualities of low resistance due to friction, long life, wide range of rotary speeds, and are shock proof. They are widely used in high precision lathes, large equipment and precision instruments. For the past 30 years, studies of hydrostatic bearings have been widely conducted. As Figure 1 shows, ordinary hydrostatic centripetal bearings use multiple oil recesses (4 to 6 oil recesses). In order to bear the load, high pressure oil channeled through the oil recesses must first be throttled. Ordinarily the throttle is a capillary throttle or an orifice throttle. Between two neighboring oil recesses, a circumferential oil groove is not recommended because when a circumferential oil groove is not present, the amount of flow required is small, and at a high speed of rotation, air flowing directly into the recesses from the oil groove because of the empty holes can be prevented. This article discusses the hydrostatic centripetal slide bearings of the capillary or orifice throttled multiple oil recess and circumferential oil groove. Analysis of this type of bearings is very complex because of the changes in the radial gaps caused by axial eccentricity, the effects of journal rotation and the flow of oil through the oil recesses. The design parameters of the bearings are many (number of oil recesses N, diameter of the shaft D, viscosity μ, velocity υ, gap c, pressure of entering oil p_s, pressure ratio $\bar{p}_n = \frac{p_n}{p_s}$, width of the axial oil sealing surface ℓ, width of the circumferential oil sealing surface ℓ_c, etc). The properties required of the bearings are also many (load capacity, stiffness, amount of flow, power loss, temperature rises, etc). They limit each other and conflict with each other. This forces us to comprehensively consider the influencing factors of all sides and use the computer to realize the optimum design. In view of the increasing speed of rotation of the shaft, heating and the rise in temperature of the bearings is a critical problem. This article discusses the optimum design with power loss as the target function under maximum load capacity and stiffness for given geometric dimensions of oil recesses.

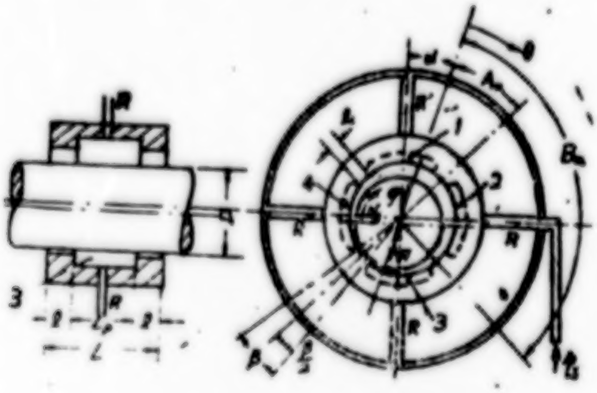


Figure 1.

II. Terms and Symbols

As shown in Figure 1, the top and bottom oil recesses are symmetric about the vertical line. The center line connecting the shaft and the bearings is the polar axis of the polar coordinates. The polar angle θ is measured clockwise.

D ----- journal diameter
L ----- length of bearings
L_p ----- length of oil recess
ℓ ----- width of axial oil sealing surface
ℓ_c ----- width of circumferential oil sealing surface
c ----- radial gap
e ----- eccentric distance
θ ----- polar angle
h ----- thickness of the oil film
$h = c(1 + \varepsilon\cos\theta)$, see equation (8)
L_t ----- length of the capillary tube
d_t ----- diameter of the capillary tube
d_o ----- diameter of orifice
K_o ----- flow coefficient of orifice
N_s ----- speed of rotation of the shaft
----- circumferential speed of the journal
N ----- total number of oil recesses
n ----- numerical order of the oil recess ($n = 1,2,\ldots\ldots.$)
α ----- angle of shift, i.e., the angle between the vertical line and the
β central line connecting the shaft and the center of the bearings
μ ----- central angle faced by the circumferential oil sealing surface
ν ----- oil viscosity
----- oil density
W ----- load of the bearings
P_s ----- pressure of oil supply
P_n ----- pressure in the nth oil recess
u ----- speed of oil flow
Q ----- amount of flow
H_p ----- power of the oil pump
H_f ----- power loss due to friction
H ----- total power loss, $H = H_p + H_f$

This article also uses the following nondimensional terms:

ε ----- ratio of eccentricity, $\varepsilon = \frac{e}{c}$
A_r ----- ratio of the area of oil recesses and total area of the bearings

$$A_r = \zeta\left(1 - \frac{2\ell}{L}\right),$$

ζ ----- ratio of the sum of the area of oil recesses (area of oil recesses + area of the circumferential oil sealing surface)

$\zeta = (1 - \frac{N}{\pi} \frac{\ell c}{D})$, see equation (40)

k ----- ratio of the width of the oil sealing surface $k = \frac{\ell c}{\ell}$

$\bar{m}$ ----- aspect ratio of the oil recesses, see equation (14)

$\bar{p}_n$ ----- pressure ratio, nondimensional, $\bar{p}_n = \frac{p_n}{p_s}$, see equation (5)

δ ----- design parameter of the capillary throttle, see equation (6)

λ ----- design parameter of the orifice throttle, see equation (7)

$\bar{\omega}$ ----- velocity parameter, see equation (15)

$\bar{W}$ ----- nondimensional load coefficient, see equation (26)

$\bar{Q}$ ----- nondimensional flow coefficient, see equation (28)

$\bar{H}$ ----- nondimensional power loss coefficient, see equation (31)

III. Hypothesis

The hypothesis used in this article is basically the same as the hypothesis used in Reference (1), summarized as follows:

1. The model of the bearings is as shown in Figure 1 (Figure 1 illustrates bearings with four oil recesses. The model of the bearings discussed in this article can have N oil recesses). All oil recesses are equal in size and are evenly distributed in the circumferential direction.

2. The fluid must not be compressed, and viscosity μ does not change at different positions on the bearings, i.e., μ = constant.

3. The flow of the liquid between the shaft and the oil sealing surface is laminar flow.

4. The depth of the oil recess is far greater than the radial gap, thus the pressure inside the oil recess can be assured to be generally even and equal.

5. The width of the circumferential oil sealing surface is extremely small compared to the length of the circumference of the oil recess. Therefore, when calculating the load capacity of the oil recess, it is assumed to be the pressure exerted upon the oil recess along the arc drawn between the centers of the oil recess and the neighboring circumferential oil sealing surface.

6. The pressure drop through each axial oil sealing surface is linear and the atmospheric pressure in the surrounding environment is regarded as zero.

7. The gaps at each point on the circumferential oil sealing surface are different and their average value is used as the average gap of the circumferential oil sealing surface.

8. Because the width of the oil sealing surface is small, the effect of dynamic pressure caused by rotation of the shaft is neglected.

9. There is no axial deviation between the shaft and the bearings, therefore the elastic deformation of the shaft and the engine base is neglected.

IV. Equation of Continuous Flow

The multiple recess hydrostatic centripetal bearings generally have four to six oil recesses. For general analysis, it is assumed that there are N evenly distributed oil recesses (Figure 1 shows only four oil recesses). Let the pressure in the nth oil recess be p_n and the pressure in the neighboring two oil recesses be p_{n-1} and p_{n+1}, where the nth oil recess extends from $\theta = A_n$ to $\theta = B_n$, then in the equation,

$$\begin{cases} A_n = \frac{\pi}{N}(2n-3) - a & (1) \\ \qquad\qquad\qquad\qquad (n = 1, 2, \ldots\ldots, N) & \\ B_n = A_n + \frac{2\pi}{N} = \frac{\pi}{N}(2n-1) - a & (2) \end{cases}$$

Also, let $p_{n-1} > p_n > p_{n+1}$, then the equation of continuous flow in the nth oil recess can be derived as follows:

(1) The amount of flow Q_1 entering the nth oil recess via the throttle when the pressure of the oil supplied by the oil pump is p_s: We know from fluid dynamics that for the capillary throttle:

$$Q_1 \text{ capillary} = \frac{\pi d_c^4}{128 L_c \mu}(p_s - p_n) = \left(\frac{Dc^3 p_s}{12\mu l}\right) \cdot \delta \cdot (1 - \bar{p}_n) \qquad (3)$$

and for the orifice throttle:

$$Q_1 \text{ orifice} = \frac{\pi}{4} K_0 d_0^2 \sqrt{\frac{2}{\rho}(p_s - p_n)} = \left(\frac{Dc^3 p_s}{12\mu l}\right) \cdot \lambda \cdot \sqrt{1 - \bar{p}_n} \qquad (4)$$

where the nondimensional pressure ratio $\bar{p}_n = \frac{p_n}{p_s}$; (5)

The nondimensional quantity

$$\delta = \left(\frac{3\pi}{32}\right)\left(\frac{d_c^4 l}{c^3 L_c D}\right) \qquad (6)$$

is called the design parameter of the capillary throttle.

The nondimensional quantity

$$\lambda=\frac{3\sqrt{2}\,\pi\mu d_0^2 K_0 l}{\sqrt{\rho p_s}\cdot c^3\cdot D} \tag{7}$$

is called the design parameter of the orifice throttle.

(2) The amount of flow Q_2 flowing from the two sides of the axial oil sealing surface when the pressure in the nth oil recess drops to p_n:

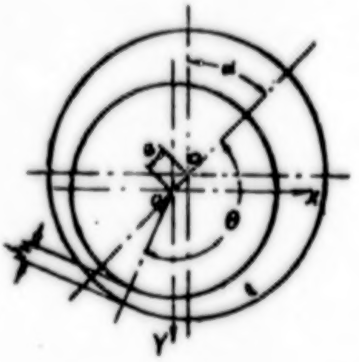


Figure 2.

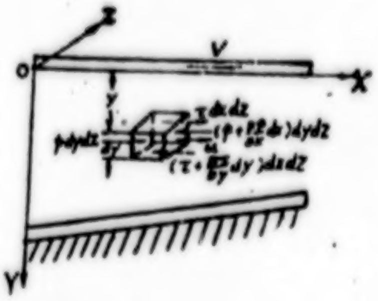


Figure 3.

As Figure 2 shows, when the shaft has an eccentricity e, the gap at any angle θ (i.e., thickness of the oil recess) is

$$h = c(1 + \varepsilon\cos\theta) \tag{8}$$

When the wedge shaped gap opens up as shown in Figure 3, it is not difficult to derive the equation for the distribution of the velocity of flow on any given cross section from the equilibrium equation of micro-elemental bodies of the oil film, Newton's law of fluid flow and boundary conditions:

$$u=\frac{1}{2\mu}\frac{\partial p}{\partial x}\cdot y(y-h)+\frac{h-y}{h}\cdot v \tag{9}$$

We can see from the above equation that the speed of flow u at y consists of two terms, the former term is caused by the pressure gradient $\frac{\partial p}{\partial x}$, the latter term is caused by velocity υ (equivalent to the circumferential velocity of the journal in the centripetal bearing). When the width in the z direction is b, the amount of flow is

$$Q=b\int_0^h u\,dy=-\frac{b}{12\mu}\frac{\partial p}{\partial x}\cdot h^3+\frac{bhv}{2} \tag{10}$$

Since $\frac{\partial p}{\partial x}$ is negative in the equation, the actual value of the amount of flow thus obtained is positive.

Based on the hypothesis (6), the pressure drop on the axial oil sealing surface is regarded as linear, i.e.,

$$\frac{\partial p}{\partial x}=\frac{0-p_n}{l}=-\frac{p_n}{l}.$$

For the axial flow on the axial oil sealing surface, the second term in equation (9) is zero. Also, the gap h at each point on the axial oil sealing surface varies with angle θ, therefore, in the nth oil recess under a pressure drop of p_n, the amount of flow from the two sides of the axial oil sealing surface is

$$\begin{aligned} Q_2&=2\int_{A_n}^{B_n}dQ=2\int_{A_n}^{B_n}\frac{\frac{D}{2}\,d\theta}{12\mu}\cdot\frac{p_n}{l}[c(1+\varepsilon\cos\theta)]^3 \\ &=\left(\frac{Dc^3p_n}{12\mu l}\right)\cdot F_n\cdot\bar{p}_n \end{aligned}\tag{11}$$

where

$$F_n=\int_{A_n}^{B_n}(1+\varepsilon\cos\theta)^3d\theta\tag{12}$$

(3) The amount of flow Q_3 flowing from the nth oil recess into the (n + 1) oil recess:

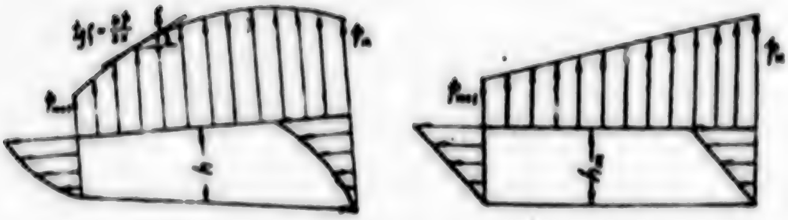
Because the pressure in the neighboring oil recess is not the same and because of the rotation of the shaft, equation (10) shows there is circumferential flow between recesses.

Because the gap h between the points on the circumferential oil sealing surface is not the same, analysis of equation (10) based on the demands of the continuity of flow shows definitely that $\frac{\partial p}{\partial x}\neq$ constant as Figure 4a shows. But the distribution of pressure cannot be easily determined. It can only be based on hypothesis (7). Taking the average value of the gap between the points as the average gap on the circumferential oil sealing surface[1] as shown in Figure 4b, we have

$$\frac{\partial p}{\partial x}=\text{constant}=\frac{p_{n+1}-p_n}{l_c}=-\frac{p_n-p_{n+1}}{l_c}$$

The average gap on the circumferential oil sealing surface is

$$h_m=\frac{1}{\beta}\int_{B_n-\frac{\beta}{2}}^{B_n+\frac{\beta}{2}}c(1+\varepsilon\cos\theta)d\theta$$



a) b)

Figure 4.

Thus, we know from equation (10) that the flow from the nth oil recess into the (n + 1) oil recess is

$$Q_s = \frac{L_p}{12\mu} \cdot \frac{p_n - p_{n+1}}{l_c} \left[\frac{1}{\beta}\int_{\theta_n - \frac{\beta}{2}}^{\theta_n + \frac{\beta}{2}} c(1+\varepsilon\cos\theta)d\theta\right]^3 + \frac{\pi N_s D L_p}{2}$$

$$\left[\frac{1}{\beta}\int_{\theta_n - \frac{\beta}{2}}^{\theta_n + \frac{\beta}{2}} c(1+\varepsilon\cos\theta)d\theta\right] = m\left(\frac{Dc^3 p_s}{12\mu l}\right)(\bar{p}_n - \bar{p}_{n+1}) \cdot H_n^3 \tag{13}$$

$$+\left(\frac{Dc^3 p_s}{12\mu l}\right)\omega\left(\frac{l}{L}\right)\left(\frac{L}{D}\right)^2\left(1 - \frac{2l}{L}\right) \cdot H_n$$

where the nondimensional quantity

$$m = \frac{\left(\frac{L}{D}\right)^2\left(\frac{l}{L}\right)\left(1 - \frac{2l}{L}\right)}{\left(\frac{l_c}{D}\right)} = \frac{L_p l}{D l_c} \tag{14}$$

is called the aspect ratio of the oil recess.

The nondimensional quantity

$$\omega = \frac{6\mu\pi N_s D^2}{c^2 \cdot p_s} \tag{15}$$

is called the velocity parameter.

The nondimensional quantity

$$H_n = \frac{1}{\beta}\int_{\theta_n - \frac{\beta}{2}}^{\theta_n + \frac{\beta}{2}} (1+\varepsilon\cos\theta)d\theta \tag{16}$$

is known from Figure 1 where $\beta = 2N\frac{l_c}{D}$ (16')

(4) Similarly, the flow Q_4 from the (n + 1) oil recess into the nth oil recess is:

$$Q_4 = \left(\frac{Dc^3 p_0}{12\mu l}\right)\Big[m(\bar{p}_{n-1} - \bar{p}_n)\cdot H^3_{n-1} + \omega\left(\frac{l}{L}\right)\left(\frac{L}{D}\right)^2\left(1 - \frac{2l}{L}\right)H_{n-1}\Big] \tag{17}$$

According to the principle of continuity of flow, the flow into the oil recess $(Q_1 + Q_4)$ = the flow from the oil recess $(Q_2 + Q_3)$, thus when the orifice is throttled:

$$\left(\frac{Dc^3 p_0}{12\mu l}\right)\cdot\lambda\cdot\sqrt{1-\bar{p}_n} + \left(\frac{Dc^3 p_0}{12\mu l}\right)\Big[m(\bar{p}_{n-1} - \bar{p}_n)H^3_{n-1} + \omega\left(\frac{l}{L}\right)\left(\frac{L}{D}\right)^2 \left(1 - \frac{2l}{L}\right)H_{n-1}\Big] = \left(\frac{Dc^3 p_0}{12\mu l}\right)\cdot F_n\cdot\bar{p}_n + \left(\frac{Dc^3 p_0}{12\mu l}\right)\Big[2m(\bar{p}_n - \bar{p}_{n+1}) \cdot H^3_n + \omega\left(\frac{l}{L}\right)\left(\frac{L}{D}\right)^2\left(1 - \frac{2l}{L}\right)H_n\Big]$$

$$\therefore \quad -C_{n-1}\bar{p}_{n-1} + (F_n + C_n + C_{n-1})\bar{p}_n - C_n\bar{p}_{n+1} = \lambda\cdot\sqrt{1-\bar{p}_n} - \omega S_n \tag{18}$$

This is the nonlinear equation for pressure p in the oil recess. In the equation F_n is determined by equation (12):

$$C_n = m\cdot H^3_n \tag{19}$$

$$S_n = \left(\frac{l}{L}\right)\left(\frac{L}{D}\right)^2\left(1 - \frac{2l}{L}\right)(H_{n-1} - H_n) \tag{20}$$

where H_n is determined by equation (16).

A comparison of equations (3) and (4) shows when the capillary tubes are throttled, we only have to change the λ to δ and the nonlinear term $\sqrt{1 - \bar{p}_n}$ to the linear term $(1 - \bar{p}_n)$ in the above equation and after combining the terms, we obtain the equation for the situation when the capillary tubes are throttled:

$$-C_{n-1}\bar{p}_{n-1} + (F_n + C_n + C_{n-1} + \delta)\bar{p}_n - C_n\bar{p}_{n+1} = \delta - \omega S_n \tag{21}$$

It is thus simplified to a linear equation.

Equations (18) and (21) are respectively the equations of continuous flow in the nth oil recess when the orifice is throttled and when the capillary tube is throttled. They are discussed in the following:

In the equations, the order of the oil recesses is n = 1,2,....N. We notice that when n = 1, $\bar{p}_{n-1} = \bar{p}_0 = \bar{p}_N$, $H_{n-1} = H_0 = H_N$, i.e., the data of the nth oil recess; when n = N, $\bar{p}_{n+1} = \bar{p}_{N+1} = \bar{p}_1$, $H_{n+1} = H_{N+1} = H_1$ i.e., the data of the first oil recess

$$\bar{p}_n = \frac{p}{p_s} \quad \text{(see equation (5))}$$

is a nondimensional quantity, the properties of the bearings are related to it, and it is an important parameter in the design of hydrostatic bearings.[2]

The aspect ratio m of the oil recess is a nondimensional quantity and is determined by equation (14). It describes the relationship of the inner dimensions of the bearings (width of the circumferential oil sealing surface and the width of the axial oil sealing surface). It can be seen in the course of derivation of equation (18) that $\frac{D}{\ell}$ affects the axial flow (see equation (11)), $\frac{L_p}{\ell_c}$ affects the circumferential flow (see equation (13)), therefore m, as a measure of the ratio of circumferential flow between neighboring oil recesses and axial flow in the oil recess, and used to clearly determine the degree of mutual effects of oil recesses, reflects the effect of circumferential flow upon the properties of the bearings, for example, when $\frac{\ell_c}{\ell}$ is small, the effect of circumferential flow is large, therefore it is another important parameter in the design of hydrostatic bearings.

The design parameters δ and λ of the throttles are nondimensional quantities determined by equations (6) and (7). They include the quantities such as the following: diameter d_t of the capillary tube (or diameter d_o of the orifice), radial gap c, pressure p_s of oil supply, viscosity μ of the oil, density ρ and some other quantities selected by the designer to facilitate optimum design. Therefore, they are design variables of hydrostatic bearings. They describe the method of the supply of oil under pressure to the oil recesses and are the bases for designing the throttles.

The velocity parameter ω is a nondimensional quantity determined by equation (15). It reflects the effects of shaft rotation.

Eccentricity ε and angle of shift α describe the position of the shaft among the bearings, therefore the coefficients F_n and H_n are only the functions of the position of the shaft among the bearings.

Thus we know that in the derivation of the equation of continuous flow in the oil recess, the series of nondimensional quantities $\bar{p}_n$, m, δ, λ, ω that appear all have obvious physical significance. They are important design variables in the design of hydrostatic bearings and directly affect the properties of the bearings.

Equation (18) or equation (21) is the general continuous equation of the nth oil recess, thus it is applicable to all oil recesses of the bearings. When we have N oil recesses and unknown oil recess pressure p_n, $(n = 1, 2, \ldots\ldots N)$, there will be N equations which can be solved simultaneously. For example, for the bearing with four oil recesses (see Figure 1), the continuous equations can be written as follows in matrix form.

When the orifice is throttled:

$$\begin{bmatrix} (F_1+C_1+C_4) & -C_1 & 0 & -C_4 \\ -C_1 & (F_2+C_2+C_1) & -C_2 & 0 \\ 0 & -C_2 & (F_3+C_3+C_2) & -C_3 \\ -C_4 & 0 & -C_3 & (F_4+C_4+C_3) \end{bmatrix} \cdot \begin{bmatrix} \bar{p}_1 \\ \bar{p}_2 \\ \bar{p}_3 \\ \bar{p}_4 \end{bmatrix} = \lambda \begin{bmatrix} \sqrt{1-\bar{p}_1} \\ \sqrt{1-\bar{p}_2} \\ \sqrt{1-\bar{p}_3} \\ \sqrt{1-\bar{p}_4} \end{bmatrix} - \omega \begin{bmatrix} S_1 \\ S_2 \\ S_3 \\ S_4 \end{bmatrix} \quad (22)$$

When the capillary tube is throttled:

$$\begin{bmatrix} (F_1+C_1+C_4+\delta) & -C_1 & 0 & -C_4 \\ -C_1 & (F_2+C_2+C_1+\delta) & -C_2 & 0 \\ 0 & -C_2 & (F_3+C_3+C_2+\delta) & -C_3 \\ -C_4 & 0 & -C_3 & (F_4+C_4+C_3+\delta) \end{bmatrix} \begin{bmatrix} \bar{p}_1 \\ \bar{p}_2 \\ \bar{p}_3 \\ \bar{p}_4 \end{bmatrix} = \begin{bmatrix} \delta-\omega S_1 \\ \delta-\omega S_2 \\ \delta-\omega S_3 \\ \delta-\omega S_4 \end{bmatrix} \quad (23)$$

Based on the above group of continuous equations, we can obtain the pressure ratio $\bar{p}_n$ of each oil recess and pressure p_n when the eccentricity $e = 0$, i.e., $\varepsilon = 0$, and the continuous equations (18) and (22) respectively degenerate to the following:

When the orifice is throttled:

$$\frac{r}{\sqrt{1-r}} = \frac{N\lambda}{2\pi} \quad (24)$$

and when the capillary tube is throttled:

$$\frac{r}{1-r} = \frac{N\delta}{2\pi} \quad (25)$$

where the pressure ratio is $r = (\bar{p}_n)_{\varepsilon = 0}$ when the eccentricity is 0 and its value can be obtained easily. It is a very important parameter in the design of bearings.

V. Load Coefficient $\bar{W}$ of the Bearings

Based on hypothesis (5), it is assumed that the oil pressure in the oil recess acts along $\overline{AB}$ (see Figure 5). Now assume that the oil pressure in the oil recess is evenly distributed along the hypotenuse $\overline{AB}$, $\overline{AB} = D \sin \frac{\pi}{N}$, then the resultant force of the oil pressure in the nth oil recess

$$P_n = p_n DL_p \sin\frac{\pi}{N}$$

∴ The angle between the y axis and the central line of the nth oil recess

$$= \left(B_n - \frac{\pi}{N} \right) + \alpha = \frac{2\pi}{N}(n-1)$$

∴ The horizontal component of p_n ,

$$(P_n)_x = P_n \sin\gamma$$
$$= P_n DL_p \sin\frac{\pi}{N} \sin\left[\frac{2\pi}{N}(n-1)\right]$$

and the vertical component

$$(P_n)_y = P_n \cos\gamma = p_n DL_p \sin\frac{\pi}{N} \cos\left[\frac{2\pi}{N}(n-1)\right]$$

Thus, the horizontal component of the total load of the entire bearing is

$$W_x = DL_p \sum_{n=1}^{N} p_n \sin\frac{\pi}{N} \sin\left[\frac{2\pi}{N}(n-1)\right]$$

The vertical component of the total load of the entire bearing is

$$W_y = DL_p \sum_{n=1}^{N} p_n \,\mathrm{sni}\, \frac{\pi}{N} \cos\left[\frac{2\pi}{N}(n-1)\right]$$

and the total load of the entire bearing is

$$W = \sqrt{W_x^2 + W_y^2} \quad .$$

When reduced to a nondimensional quantity, the defined load coefficient $\overline{W}$ is the ratio of the total load capacity of the bearing and the load capacity under pressure p_s of the oil supply when the area filled with oil is DL_p. Thus

$$\overline{W} = \frac{W}{p_s DL_p} = \sqrt{\overline{W}_x^2 + \overline{W}_y^2} \qquad (26)$$

where
$$\overline{W}_x = \sum_{n=1}^{N} \overline{p}_n \sin\frac{\pi}{N} \sin\left[\frac{2\pi}{N}(n-1)\right],$$
$$\overline{W}_y = \sum_{n=1}^{N} \overline{p}_n \sin\frac{\pi}{N} \cos\left[\frac{2\pi}{N}(n-1)\right]$$

and in Figure 1, the defined load angle

$$\varphi = \mathrm{tg}^{-1}\frac{W_x}{W_y} \tag{27}$$

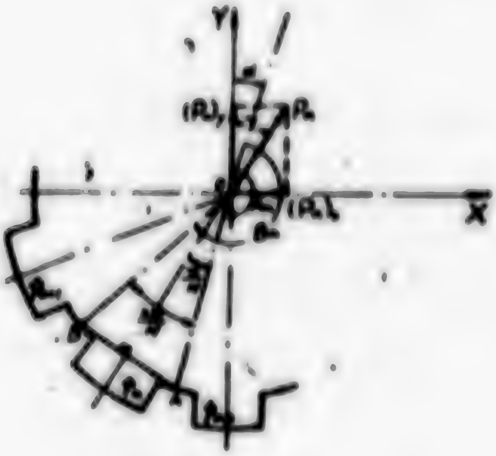


Figure 5.

VI. Flow Coefficient $\bar{Q}$ of the Bearing

To the entire bearing, the flow among the oil recesses is the internal flow among the bearings and the total flow through all the bearings is

$$Q = \sum_{n=1}^{N} Q_n$$

where Q_n --- the flow in each throttle.

For the capillary throttle:

We know from equation (3)

$$Q = \frac{Dc^3 p_s}{12\mu l}\,\delta \sum_{n=1}^{N}(1-\bar{\beta}_n) = \left(\frac{2\pi D c^3 p_s}{12\mu l}\right)\cdot\frac{\delta}{2\pi}\cdot\sum_{n=1}^{N}(1-\bar{\beta}_n)$$

For the orifice throttle:

We know from equation (4)

$$Q = \frac{Dc^3 p_s}{12\mu l}\cdot\lambda\cdot\sum_{n=1}^{N}\sqrt{1-\bar{\beta}_n} = \left(\frac{2\pi D c^3 p_s}{12\mu l}\right)\cdot\frac{\lambda}{2\pi}\cdot\sum_{n=1}^{N}\sqrt{1-\bar{\beta}_n}$$

When reduced to a nondimensional quantity, the defined flow coefficient $\bar{Q}$ of the bearing is the ratio of the total flow through the bearings and the flow through the bearings when the eccentricity is zero and the pressure in all of the oil recesses is equal to the pressure p_s of the oil supply, i.e.,

$$\bar{Q} = \frac{Q}{2\,\frac{\pi D c^3 p_s}{12\mu l}} \tag{28}$$

∴ For the capillary throttle:

$$\bar{Q} = \frac{\delta}{2\pi}\sum_{m=1}^{N}(1-\bar{p}_m) \tag{29}$$

For the orifice throttle:

$$\bar{Q} = \frac{\lambda}{2\pi}\sum_{m=1}^{N}\sqrt{1-\bar{p}_m} \tag{30}$$

When the eccentricity is zero, equation (29) becomes

$$\bar{Q} = \frac{\delta}{2\pi}\cdot N\cdot(1-r)$$

, and substituting equation (25) into it we get $\bar{Q} = r$; and equation (30) becomes

$$\bar{Q} = \frac{\lambda}{2\pi}\cdot N\cdot\sqrt{1-r}$$

, substituting equation (24) into it we get $\bar{Q} = 4$.

VII. Power Loss Coefficient $\bar{H}$

The total power H lost by the hydrostatic bearing is equal to the sum of the power H_p needed by the pump to supply the bearings with pressured oil and the H_f lost due to the fluid's viscous friction produced by rotation of the shaft around the bearings. The latter includes three parts: (1) friction loss on the axial oil sealing surface; (2) friction loss on the circumferential oil sealing surface; (3) friction loss in the oil recess.

The defined power loss coefficient $\bar{H}$ of the hydrostatic bearing is the ratio of the total power loss in the bearing and the power H_p of the pump needed to supply the bearing with pressured oil when the eccentricity is zero,

i.e.,

$$\bar{H} = \frac{H_p + H_f}{H_p} = 1 + \frac{H_f}{H_p} = 1 + \bar{H}_f \tag{31}$$

The power of the pump needed to supply the bearing with pressured oil is

$$H_p = p_s \cdot Q = \frac{\pi D c^3 p_s^2}{6 \mu l} \bar{Q} \tag{32}$$

From equation (9) we know the shearing stress between oil layers is

$$\tau = -\mu \frac{du}{dy} = \frac{1}{2} \frac{\partial p}{\partial x} (h - 2y) + \frac{\mu V}{h}$$

At the oil journal, y = 0,

$$\tau \Big|_{y=0} = \frac{h}{2} \frac{\partial p}{\partial x} + \frac{\mu V}{h} \tag{33}$$

and when $\frac{\partial p}{\partial x} = 0, \quad \tau \Big|_{y=0} = \frac{\mu V}{h}$ (34)

(1) The friction power loss H_{f1} on the axial oil sealing surface:

The average gap on the axial oil sealing surface $= \frac{1}{2\pi} \int_0^{2\pi} c(1 + \varepsilon \cos\theta) d\theta = c$,

to the axial oil sealing surface, the dimension is very short. The circumferential pressure gradient caused by the effects of dynamic pressure can be small enough and neglected, i.e., we can assume that the circumferential pressure gradient $\frac{\partial p}{\partial x} = 0$, therefore from equation (34) we know that the shearing stress at the journal on the axial oil sealing surface is $\tau_{xa} = \frac{\mu V}{c}$, and the area of the axial oil sealing surface is $A_a = 2\pi D\ell$, $V = \pi N_s D$. The friction power loss on the axial oil sealing surface is

$$H_{f1} = \frac{2\pi^3 \mu l D^3 N_s^2}{c} = K_1 \cdot \left(\frac{\pi D c^3 p_s^2 \bar{Q}}{6 \mu l} \right)$$

where

$$K_1 = \frac{1}{3\bar{Q}} \left(\frac{l\omega}{D} \right)^2 = \frac{\omega^2}{3\bar{Q}} \left(\frac{l}{L} \right)^2 \left(\frac{L}{D} \right)^2 \tag{35}$$

(2) Friction power loss H_{f2} on the circumferential oil sealing surface:

On the circumferential oil sealing surface, according to hypothesis (7), the average value of the gaps at all points is used as that of the average gap, therefore the average gap on the circumferential oil sealing surface behind the nth oil recess is

$$h_n' = \frac{c}{\beta}\int_{\theta_n-\frac{\beta}{2}}^{\theta_n+\frac{\beta}{2}} (1+\varepsilon\cos\theta)\,d\theta = cH_n$$

(H_n is determined by (16)). At this time, we can assume that the circumferential pressure gradient is

$$\frac{\partial p}{\partial x} = \text{constant} = \frac{p_{n+1}-p_n}{l_c},$$

then we know from equation (33), the shearing stress at the journal on the circumferential oil sealing surface behind the nth oil recess is

$$\tau_{x1} = \frac{h_n}{2}\cdot\frac{p_{n+1}-p_n}{l_c} + \frac{\mu}{h_n}\cdot V$$

∴ The area of each circumferential oil sealing surface is $A_1 = L_p \cdot l_c = L_p\left(\frac{D}{2}\beta\right)$, and its friction resistance is $(T_1)_n = \tau_{x1} \cdot A_1$, then from equation (15), we can represent $V(=\pi N_s D)$ by ω, thus the power loss due to friction on the $^{*}N$ circumferential oil sealing surface is

$$H_{f2} = \sum_{n=1}^{N}(T_1)_n\cdot V = \left(\frac{\pi D c^3 p_s^2 \bar{Q}}{6\mu l}\right)\sum_{n=1}^{N}\left[K_1\cdot H_n\cdot(\bar{p}_{n+1}-\bar{p}_n) + \frac{K_2}{H_n}\right]$$

where

$$K_1 = \frac{\beta m \omega}{4\pi\bar{Q}} \tag{36}$$

$$K_2 = \frac{m l_c \beta\omega^2}{12\pi D\bar{Q}} = \frac{\omega^2\beta}{17\pi\bar{Q}}\cdot\left(\frac{l}{L}\right)\cdot\left(\frac{L}{D}\right)^2\cdot\left(1-\frac{2l}{L}\right) \tag{37}$$

(3) Power loss due to friction in the oil recess:

For the hydrostatic bearing of low speed and deep oil recesses, the friction loss in the oil recesses can be neglected, but for a hydrostatic bearing of high speed, the friction loss in the oil recesses cannot be neglected.

In the oil recesses, the moment of resistance due to friction resulting from the shearing of oil is caused by the pure viscous shearing stress when the flow in the oil recesses is laminar and it is caused by the shearing stress of more complex turbulence and vortical movements when the velocity has surpassed the scope of laminar flow. Here, based on hypothesis (3), discussion will be limited to within the scope of laminar flow. Because the oil recesses are deeper (in general, the depths of the oil recesses are $h_p > 50c$, c --radial gap) and based on hypothesis (4), we can regard the pressure in the oil recesses as generally even and equal, i.e., $\frac{\partial p}{\partial x} = 0$,[3] therefore, from equation (34) we know the shearing stress at the journal of the oil recess is

$$\tau_{sp} = \frac{\mu V}{h_p} \tag{38}$$

Because the depth h_p of the oil recess is far larger than the eccentricity, we can regard the friction resistance in all oil recesses as generally equal, thus the power loss due to friction in the oil recesses is

$$H_{f3} = \frac{\pi\zeta\mu D L_p V^2}{h_p} = K_4 \cdot \left(\frac{\pi D c^3 p_s^2 \overline{Q}}{6\mu l}\right)$$

where $$K_4 = \frac{\zeta L_p l c \omega^2}{6 h_p D^2 \overline{Q}}$$

$$\zeta = \frac{\text{area of oil recesses}}{\text{area of oil recesses + area of circumferential oil sealing surface}}$$

$$= 1 - \frac{N l_c}{\pi D} = \frac{l_p}{l_p + l_c} \tag{40}$$

It can be seen from the above equation that when the velocity is not high and when the oil recess is deeper, the power loss due to friction in the oil recesses is relatively small and can be neglected.

The viscous friction power loss of the entire bearing is $H_f = H_{f1} + H_{f2} + H_{f3}$, reduced to a nondimensional quantity,

$$\overline{H}_f = \frac{H_f}{\left(\frac{\pi D c^3 p_s^2 \overline{Q}}{6\mu l}\right)}\left[K_1 + K_2 \cdot \sum_{m=1}^{N} H_m(\overline{p}_{m+1} - \overline{p}_m) + K_3 \cdot \sum_{m=1}^{N} \frac{1}{H_m} + K_4\right] \tag{41}$$

Thus from equation (31) we know that the power loss coefficient of the entire bearing is $\overline{H} = 1 + \overline{H}_f$.

It can be seen from equation (32) and equation (41) and equations (35), (36), (37), (39) and (15) that power loss is mainly determined by the radial gap c and the velocity parameter ω. The total power loss increases as the speed of rotation increases. In the low speed zone, the main power loss is the pump's power while in high speed zones, the major power loss is power loss due to viscous friction.

VIII. Analysis and Discussion

Based on the above computational formulas, the relationship between the series of properties ($\bar{W}$, $\bar{Q}$, $\bar{H}$) of the multiple recess hydrostatic bearing and changes in the design parameters (m, λ or δ, ω, $\bar{p}_n$ or r) is as follows.

Because there are too many parameters in the design of the bearings and to be able to draw the curves on a plane, only a few parameters can be fixed to observe the effects of the variations of the other parameters upon the properties of the bearing.

(1) The design parameters λ or δ of the throttles:

The curves of variation of the load coefficient $\bar{W}$ along with changes in the design parameters λ or δ are illustrated in Figure 6 and Figure 7 respectively.[4] In the figures, each curve corresponds to a different eccentricity value ε. It can be seen from the figures that when there is no eccentricity ($\varepsilon = 0$), the pressure in each oil recess is equal, and obviously the bearing cannot bear any load. The greater the ε, the greater the load capacity but it is limited by the smallest thickness of the oil film h_{min} ($h_{min} = c\,(1 - \varepsilon)$). When a high speed hydrostatic bearing is used in a precision lathe (such as a grinder) to achieve processing precision as the main objective, ε must be small. For a given value of ε and when the design parameter λ (or δ) is of a definite value, a greatest load coefficient exists. It can be seen from equations (26) and (18) and (22) (or equations (21) and (23)) that the load coefficient $\bar{W}$ is a monotonic continuous function of λ (or δ).

$$\bar{W} = f(\lambda)[\text{ or } \bar{W} = f(\delta)],$$

From $\frac{\partial \bar{W}}{\partial \lambda} = 0$ (or $\frac{\partial \bar{W}}{\partial \delta} = 0$) we can solve the value of λ at $\bar{W}_{max}$ (or the value of δ).

The curves of the flow coefficient $\bar{Q}$ as it varies with the changes in the design parameters λ or δ are shown in Figure 8 and Figure 9.

(2) The aspect ratio m of the oil recess: As shown in Figures 6 and 7, when $m \to 0$, the width of the circumferential sealing surface ℓ_c is very large, there is minimal circumferential flow, $\therefore m = 0$ can be regarded as indicating no circumferential flow, thus the pressure in each oil recess is calculated on the basis that all the oil flow only from the axial oil sealing surface. When m increases, more oil flows from the oil recesses of high pressure to the neighboring oil recesses of lower pressure, leading to a drop in the pressure in the oil recesses and the load capacity drops, as shown in Figure 6 and Figure 7.

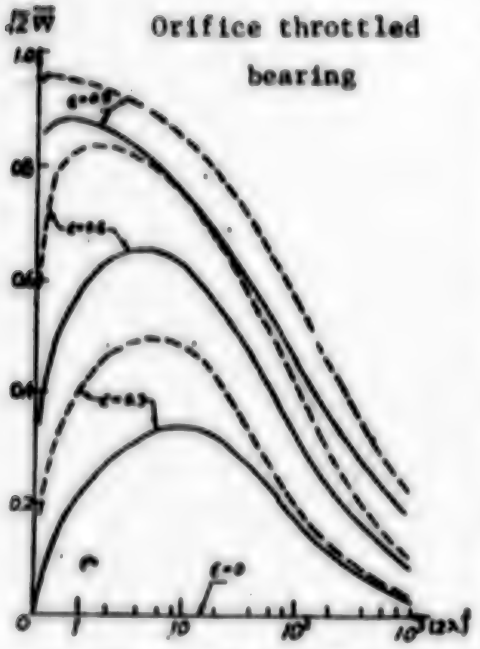


Figure 6

(Dotted line m = 0; i.e., there is no circumferential flow; solid line m = 0.77)

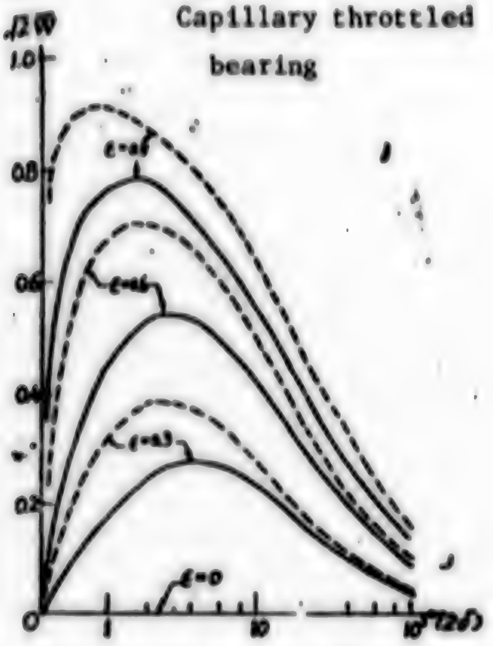


Figure 7

(Dotted line m = 0; i.e, there is no circumferential flow, solid line m = 0.77)

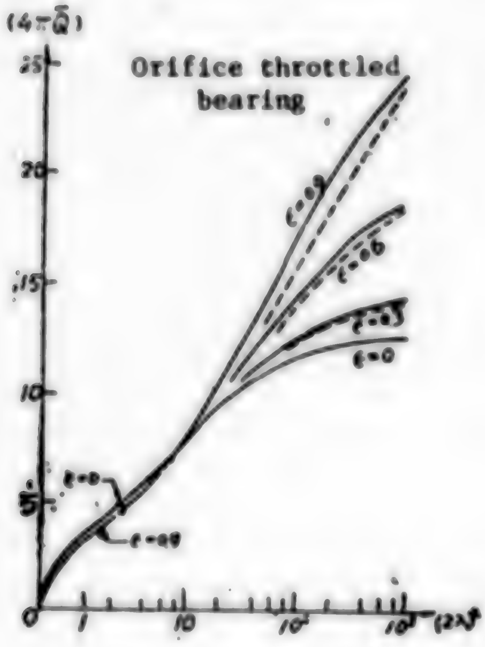


Figure 8

(Dotted line m = 0; i.e., there is no circumferential flow, solid line m = 0.77)

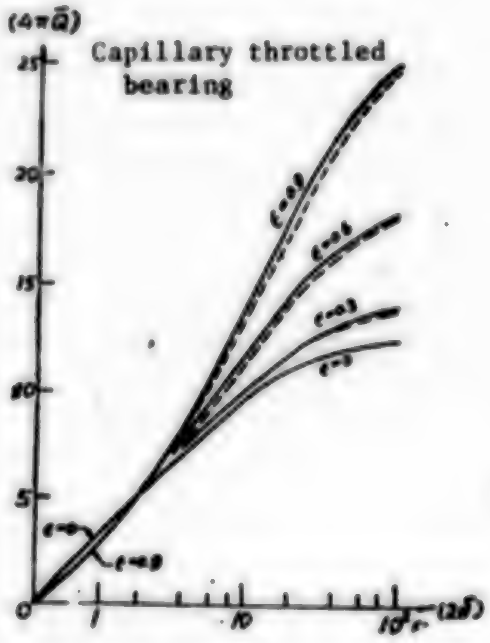


Figure 9

(Dotted line m = 0; i.e., there is no circumferential flow; solid line m = 0.77)

This shows that when m is large, the effect of circumferential flow upon the load capacity cannot be neglected. We know from equation (14) that the method of increasing m is to reduce the width ℓ_c of the circumferential oil sealing surface between neighboring oil recesses, increase the width ℓ of the axial oil sealing surface, increase the axial length L_p of the oil recess or reduce the diameter D of the shaft.

It can be seen in Figure 8 and Figure 9 that m does not exert a great effect upon the flow coefficient $\bar{Q}$. This is because m affects only the ratio of circumferential flow and axial flow in the oil recess. But to the entire bearing, circumferential flow is internal flow and it basically does not affect total flow.

(3) Velocity parameter ω:

There are two causes of circumferential flow: (1) pressure differential of neighboring oil recesses, and (2) velocity of rotation of the shaft.

Figures 10, 11, 12 and Figures 13, 14, 15[5] respectively show the curves of the load W, flow Q and power loss H of the capillary throttle bearings as they vary with the velocity parameter ω.

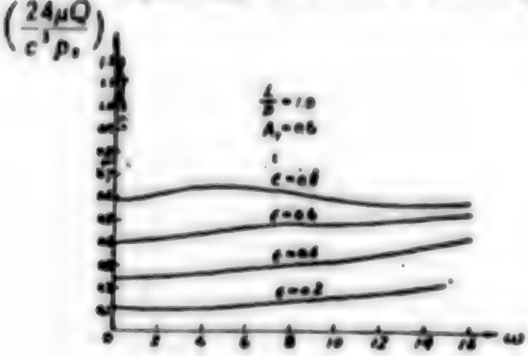

Capillary throttled bearing

Figure 10

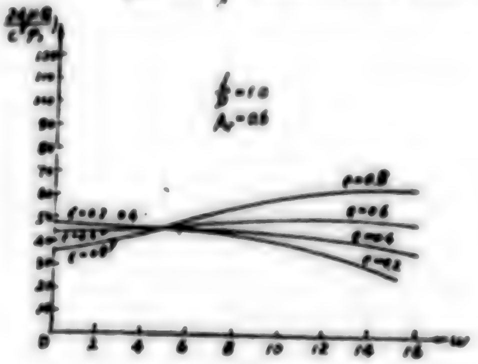

Capillary throttled bearing

Figure 11

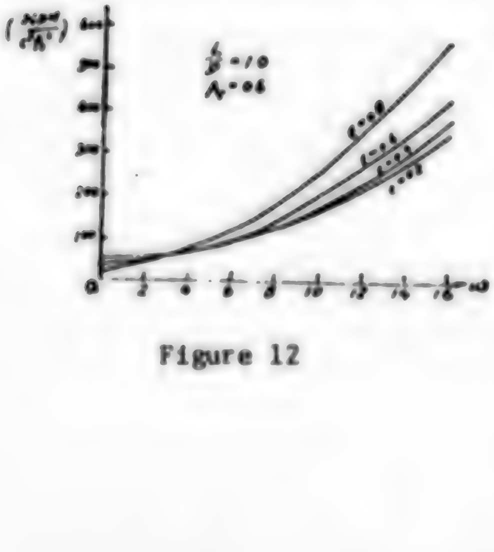

Capillary throttled bearing

Figure 12

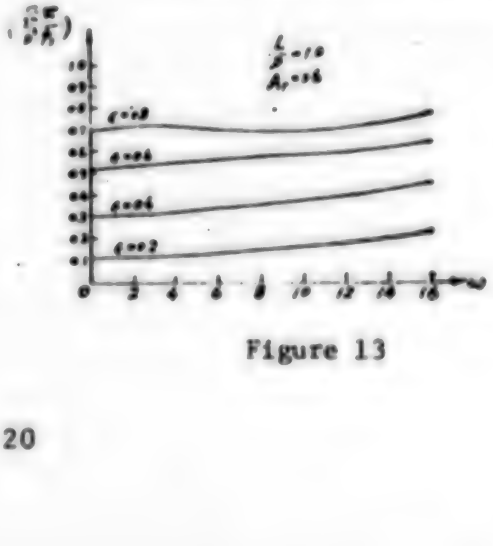

Orifice throttled bearing

Figure 13

Orifice throttled bearing

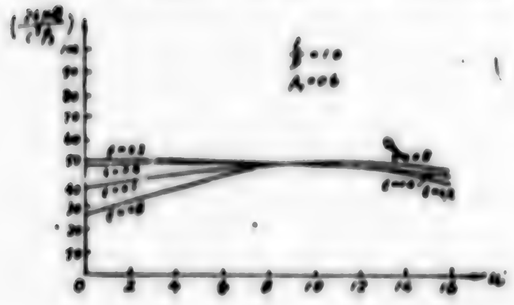


Figure 14

Orifice throttled bearing

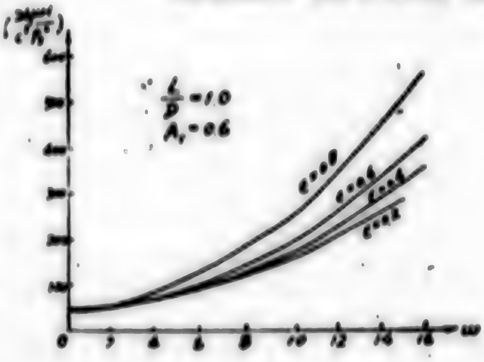


Figure 15

Figures 12 and 15 show the total power loss H increases as the speed of rotation increases. Figures 11 and 14 show the variation of the power loss H_p of the pump is small, thus the power loss due to viscous friction H_f increases ($\therefore H_f = H - H_p$). This shows that in the low velocity zone, the major power loss is the power of the pump while in the high velocity zone, it is mainly loss due to viscous friction.

The bearing can be designed according to these diagrams, for example, based on the load W, the eccentricity ε, aspect ratio of the oil recess m, and the design parameter λ or δ of the throttles can be determined from Figure 6 or Figure 7; the flow Q can be solved from Figure 8 or Figure 9; the velocity parameter ω can be determined from Figure 10 or Figure 13; and the power loss H can be solved from Figure 12 or Figure 15. But we must note that the prerequisite of Figures 10-15 is: $\frac{L}{D} = 1.0$; $A_r = 0.6$. The following computer program for optimum design can provide the best data and design properties for the design parameters of the bearings under any initial condition.

IX. Optimum Design of the Hydrostatic Bearing

By summarizing the above based on the hypothetical model of the bearing and by writing down the equations of continuity of flow and equations for energy loss, we obtain a series of computational formulas. For a specific fluid hydrostatic centripetal bearing, $\bar{p}_n$ can first be computed from equations (22) or (23), then $\bar{W}$, $\bar{Q}$ and $\bar{H}$ can be computed from equations (26) and (28) or equations (29) and (31). The key to this analysis is to solve the nondimensional pressure $\bar{p}_n$ of the oil recesses. For the capillary throttled bearing, these values can be obtained directly on the computer by solving the linear equations (23) using the inverse matrix method.

Now we use the capillary throttled four oil recess hydrostatic centripetal bearing as an example. The matrix shown in equation (23) can be simplified as

$$AX = B$$

where the matrix of the coefficients is

$$A = \begin{bmatrix} (F_1 + C_1 + C_4 + \delta) & -C_1 & 0 & -C_4 \\ -C_1 & (F_2 + C_2 + C_1 + \delta) & -C_2 & 0 \\ 0 & -C_2 & (F_3 + C_3 + C_2 + \delta) & -C_3 \\ -C_4 & 0 & -C_3 & (F_4 + C_3 + C_4 + \delta) \end{bmatrix}$$

The unknown column matrix is

$$X = \begin{bmatrix} \bar{p}_1 \\ \bar{p}_2 \\ \bar{p}_3 \\ \bar{p}_4 \end{bmatrix}$$

The constant column matrix is

$$B = \begin{bmatrix} \delta - \omega S_1 \\ \delta - \omega S_2 \\ \delta - \omega S_3 \\ \delta - \omega S_4 \end{bmatrix}$$

When we use the standard subroutine for computing the inverse of the matrix, we can obtain the inverse matrix A^{-1} of the coefficient matrix A, thus the solution of the linear equations is

$$X = A^{-1}B$$

To increase the precision, the program flow chart illustrated in Figure 16 can be used.

For the orifice throttled bearing, the damping least square method (Marquardt method) can be used.

As mentioned previously, since the bearing has many design parameters and there are many properties of the bearings that have to be realized and they mutually limit each other, thus the influencing factors on all sides must be comprehensively considered and the computer is used to realize the optimum design. In view of the trend towards faster speed of rotation, heating of the bearing and the rise in temperature are a critical problem. Therefore, this article's discussion of optimum design of the bearing takes power loss as the target function. In the design, it is always hoped that the load capacity can be greater and the stiffness of the bearing can be higher, therefore for given dimensions of oil recesses, the maximum load capacity and maximum stiffness[6] are taken as the constraints. (If the given load is known, the constraining conditions would be that the load capacity of the oil film must equal the external load.) In addition,

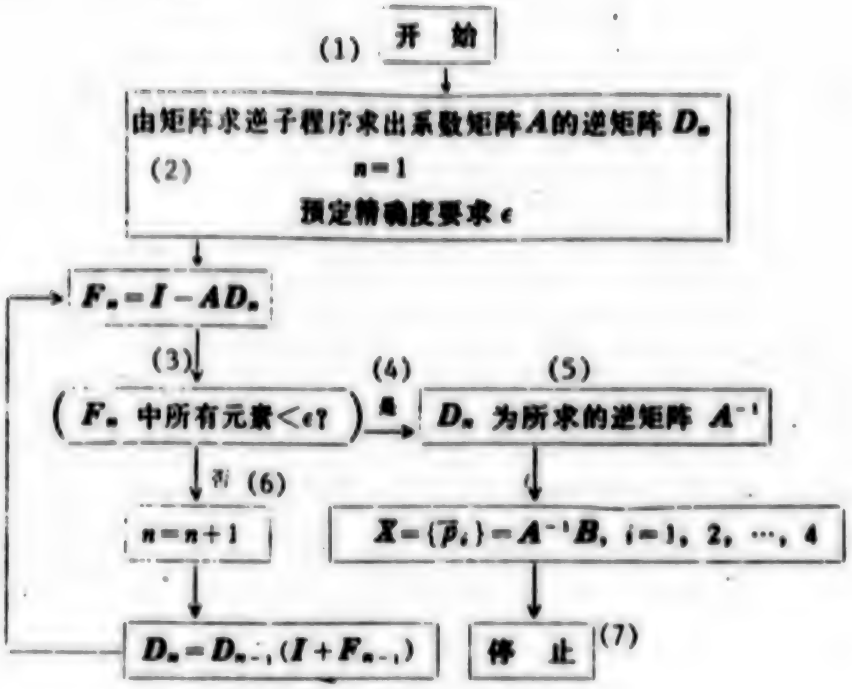


Figure 16

Key:
(1) Start
(2) Using the subroutine to obtain the inverse of the matrix to solve the inverse matrix D_n of the coefficient matrix A $n = 1$

Predetermined precision required is ε

(3) All elements in $F_n < \varepsilon$?
(4) Yet
(5) D_n is the inverse matrix A^{-1} sought
(6) No
(7) Stop

because of the limitations of the hypothesis used in this article, the ratio of the total area of the oil recesses and the total area of the bearings cannot be less than 0.5. Otherwise, it means the area of the oil sealing surface relative to the area of the oil recesses is too large and these assumptions will be incorrect. Therefore, we let $A_r > 0.5$ be another constraining condition. Some of the bearing parameters can be given beforehand. Parameters that do not change in optimum design are called predetermined parameters. The parameters that directly affect the load and power loss are considered as variable design variables. In the design of the hydrostatic bearing, the speed of rotation N_s of the shaft often has already been determined in the total design of the machinery. Some other parameters such as pressure p_s of the oil supply, radial gap c, eccentricity ε, ratio of length and diameter $\frac{L}{D}$, viscosity μ and the number of oil recesses N are frequently all given beforehand by the predetermined parameters. For example, in a high speed grinder, the main goal is to satisfy processing precision and surface smoothness, and the load capacity is not important. At this time, the eccentricity ε should be given beforehand according to the precision demands. In this way, equations (26) and equations (18), (21,) (22), and (23) show that the load coefficient $\bar{W}$ is related to $\frac{\ell_c}{D}$, $\frac{\ell}{L}$ and λ (or δ). In addition, equations (48) and equations (35), (36), (37) and (47) show the power loss coefficient $\bar{H}$ is also related to $\frac{\ell c}{D}$, $\frac{\ell}{L}$ and λ (or δ). In fact, $\frac{\ell c}{D}$ and $\frac{\ell}{L}$ reflect the effect of the oil sealing surface, λ or (δ) reflects the effect of the throttle. Therefore, $\frac{c}{D}$, $\frac{\ell}{L}$ and λ (or δ) are considered as the design variables. The intermediate variables to be used (working variables) are $\bar{W}$, $\bar{Q}$, $\bar{H}$, a, p_1, p_2, p_3 are functions of the design variables and the predetermined parameters. Therefore, the mathematical model of the optimum design problem of the fluid hydrostatic centripetal bearing is:

Solving the design variable $\quad \boldsymbol{X}=\left\{\frac{l_c}{D}\ \frac{l}{L}\ \ \lambda(\text{or}\,\delta)\right\}$ (42)

so that the target function $\quad f(\boldsymbol{X})=\bar{H}_{min}$

They satisfy the restraining conditions:

$$\left.\begin{array}{ll} g_1(\boldsymbol{X})=\dfrac{\partial\bar{W}}{\partial\lambda(\text{or}\,\partial\delta)}=0 & \text{(or the external load } P - W = 0 \\ g_2(\boldsymbol{X})=A_r-\left(1-\dfrac{N}{\pi}\cdot\dfrac{l_c}{D}\right)\left(1-\dfrac{2l}{L}\right)<0 & \text{(}A_r\text{ can be selected to be } 0.5 \end{array}\right] \quad (43)$$

The remaining restraining conditions are

$$\left.\begin{array}{c} 0<\dfrac{l_c}{D}<\dfrac{\pi}{N} \\ 0<\dfrac{l}{L}<\dfrac{1}{2} \\ 0<\lambda\ (\text{or}\,\delta) \end{array}\right\} \quad (44)$$

Because the target function and the restraining conditions are nonlinear functions of the design variables, this is a nonlinear programming problem with constraints. In the optimum design of hydrostatic bearing discussed in this article, the expressions of the target function and the constraining conditions are more complex, and it is difficult to compute the derivatives. But they are still quadratic type functions, therefore it was decided to use the penalty function method and Powell's method which does not require the computation of derivatives for computing the optimum design. To reduce the problem of optimization with constraints (equations (42)-(44)) to a series of problems of optimization without constraints, we can construct a new penalty function F(X) based on the quadratic type conditions of constraints (43) as follows:

Let

$$g_i^+(\boldsymbol{X}) = (g_i(\boldsymbol{X}))^2$$

$$g_k^+(\boldsymbol{X}) = \frac{1}{2}[\,|g_k(\boldsymbol{X})|\,g_k + (\boldsymbol{X})\,]$$

then $$F(\boldsymbol{X}) = f(\boldsymbol{X}) + M\cdot[g_i^+(\boldsymbol{X}) + g_k^+(\boldsymbol{X})] \qquad (45)$$

where the penalty factor M in the equation is a large positive number. In this way, the problem of optimum design with constraints expressed by equations (42) and (43) becomes a problem of solving a series of unconstrained functions F(X) for their unconditional minima:

Solving the design variable

$$\boldsymbol{X} = \left\{ \frac{l_c}{D} \;\; \frac{l}{L} \;\; \lambda \text{ (or } \delta) \right\}$$

so that the unconstrained target function (F(X), M_k) is minimum, while

$$0 < M_1 < M_2 < \cdots\cdots < M_{K-1} < M_K < M_{K+1} < \cdots\cdots$$

$$\lim_{K\to\infty} M_K = \infty, \qquad (46)$$

We first begin by searching the outer part of the feasibility range. When we are far away from the border of the feasibility range, M_k is relatively small. As M is taken as larger positive numbers, the solution of unconditional extrema $X = X(M_k)$ of the corresponding series of penalty functions (F(X, M_k)) becomes closer and closer to the feasibility range. When M is taken to be sufficiently large (→ ∞), the solution X_k (=X^*) is the optimum solution obtained by gradually approaching the border of the feasibility range from the outside, i.e., the optimum solution of the original hydrostatic bearing design. Its program flow chart is shown in Figure 17.

Because the expression (45) of the penalty function (F(X)) is complex, calculating the derivatives is difficult, therefore a direct search method is used--Powell's method.[8] Its program flow chart is shown in Figure 18.

The one dimensional search in the program flow chart (Figure 18) of Powell's method used the 0.618 method. Its program flow chart can be located in Reference (11).

The design equations and optimum design programs combined in Figures 16-18 described above provide the method of computation for the optimum design of the design variables $\frac{\ell c}{D}$, $\frac{\ell}{L}$ and λ (or δ) using the computer, and taking power loss as the target function and the load capacity and stiffness as constraints. The FORTRAN program for the design of the four recess hydrostatic centripetal bearing has been written and compiled according to this flow chart. Based on this program, the best data for the rational selection of dimensions of the oil recesses and the design parameters of the throttles in the design of hydrostatic bearings can be obtained.

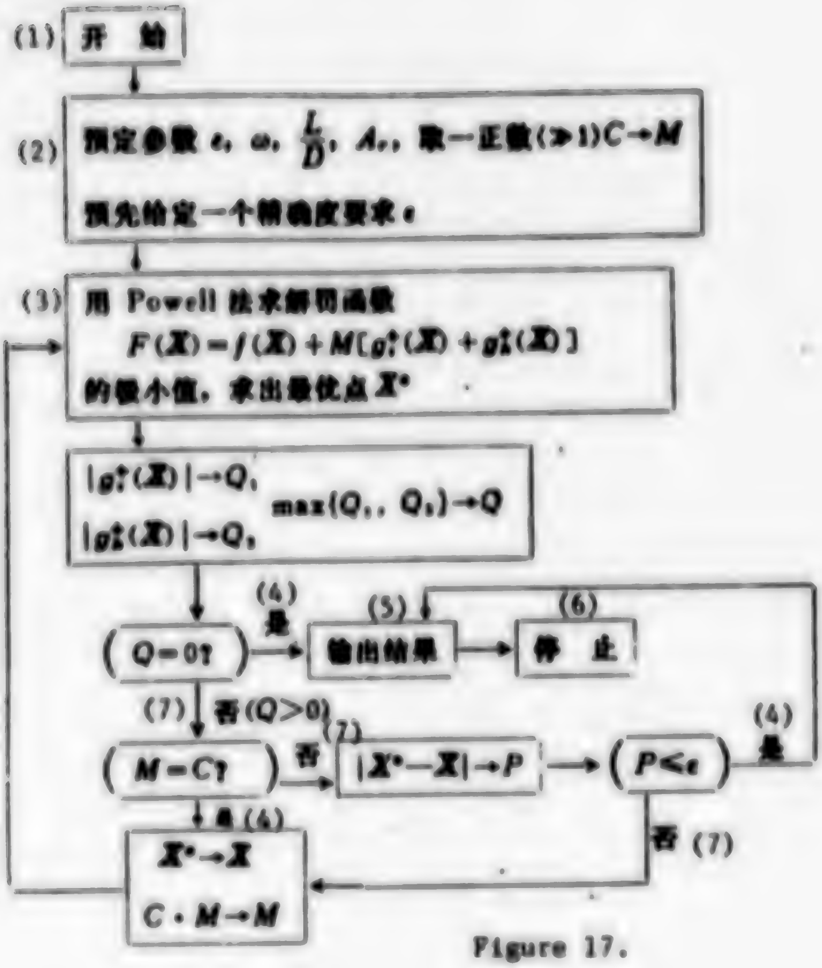


Figure 17.

Key:

(1) Start

(2) Predetermined parameters ε, ω, $\frac{L}{D}$, A_r, take a positive number (>> 1) C →M
Given precision demand   beforehand

(3) Use Powell's method to solve the minimum value of the penalty function $F(X) = f(X) + M[g_1^t(X) + g_h^t(X)]$ to find the optimum point X*.

(4) Yes

(5) Output result

(6) Stop

(7) No

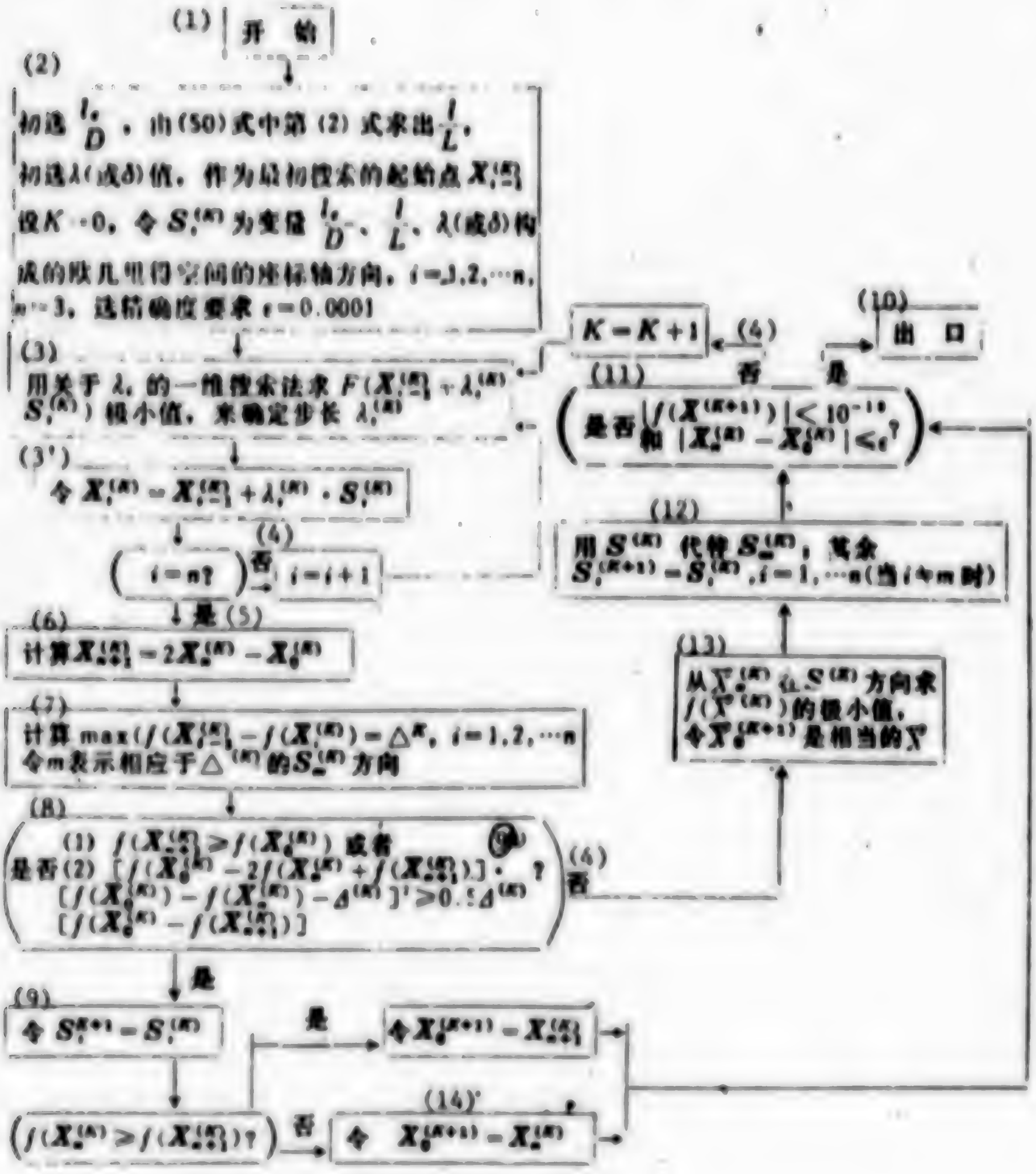

(1) 开 始
(2) 初选 lc/D，由(50)式中第(2)式求出 l/L，初选λ(或δ)值，作为最初搜索的起始点 X(K)i-1 设K=0，令 S(K)i 为变量 lc/D、l/L、λ(或δ)构成的欧几里得空间的坐标轴方向，i=1,2,…n，n=3，选精确度要求 ε=0.0001
(3) 用关于λi的一维搜索法求 F(X(K)i-1 + λ(K)i S(K)i) 极小值，来确定步长 λ(K)i
(3') 令 X(K)i = X(K)i-1 + λ(K)i · S(K)i
i=n?
否
(4) i=i+1
是 (5)
(6) 计算 X(K)n+1 = 2X(K)n − X(K)0
(7) 计算 max(f(X(K)i-1) − f(X(K)i)) = △K，i=1,2,…n 令m表示相应于△(K)的S(K)m方向
(8) 是否 (1) f(X(K)n+1) ≥ f(X(K)0) 或者 (2) [f(X(K)0) − 2f(X(K)n) + f(X(K)n+1)]·[f(X(K)0) − f(X(K)n) − Δ(K)]² ≥ 0.5Δ(K)[f(X(K)0) − f(X(K)n+1)]² ?
(6) 否
是
(9) 令 S(K+1)i = S(K)i
(f(X(K)n) ≥ f(X(K)n+1))?
是
令 X(K+1)0 = X(K)n+1
否
(14)'
令 X(K+1)0 = X(K)n
K=K+1
(4)
(10) 出 口
否
是
(11) 是否 |f(X(K+1))| ≤ 10⁻¹⁰ 和 |X(K)n − X(K)0| < ε ?
(12) 用S(K)代替S(K)m，其余 S(K+1)i = S(K)i，i=1,…n(当i≠m时)
(13) 从X(K)n沿S(K)方向求f(X(K))的极小值，令X(K+1)0是相当的X

Figure 18

Key:

(1) Start

(2) First select $\frac{\ell c}{D}$, solve $\frac{\ell}{L}$ from equation (2) in equations (50), first select the value of λ (or δ) as the starting point for the initial search $X_{i-1}^{(K)}$

Let K = 0, let $S_i^{(K)}$ be the direction of the coordinate axis of the Euclidean space constructed by the variables $\frac{\ell c}{D}$, $\frac{\ell}{L}$, λ (or δ), i = 1,2,........n, n = 3, the precision required is = 0.0001

[Key to Figure 18 continued]

(3) Use the one dimensional search for λ_i to obtain the minima of $F(X_{i-1}^{(K)} + \lambda_i^{(K)} S_i^{(K)}$ and determine the step length $\lambda_i^{(K)}$

(3') Let $X_i^{(K)} = X_{i-1}^{(K)} + \lambda_i^{(K)} \cdot S_i^{(K)}$

(4) No

(5) Yes

(6) Compute $X_{n+1}^{(K)} = 2X_n^{(K)} - X_0^{(K)}$

(7) Compute $_{\max} f(X_{i-1}^{(K)} - f(X_i^{(K)}) = \Delta^K$, $i = 1,2,\ldots n$

Let m represent the $S_m^{(K)}$ direction of $\Delta^{(K)}$

(8) Whether (1) $f(X_{n+1}^{(K)} \geq f(X_0^{(K)}$

(2) $[f(X_0^{(K)} - 2f(X_n^{(K)} + f(X_{n+1}^{(K)})]\cdot$?

$[f(X_0^{(K)}) - f(X_n^{(K)}) - \Delta^{(K)}]^2 \geq 0.5\ \Delta^{(K)}$

$[f(X_0^{(K)} - f(X_{n+1}^{(K)})]$

(9) Let $S_i^{K-1} = S_i^{(K)}$

(10) Output

(11) Whether $\begin{pmatrix} |f(X^{(K+1)})| < 10^{-10} \\ |X_n^{(K)} - X_0^{(K)}| \ \epsilon_c \end{pmatrix}$?

(12) Substitute $S^{(K)}$ for $S_m^{(K)}$, the remaining $S_i^{(K+1)} = S_i^{(K)}$, $i = 1, \ldots n$

(when $i \neq m$)

(13) Solve the minima of $f(\vec{X}^{(K)})$ from $\vec{X}_n^{(K)}$ in the $S^{(K)}$ direction and let $\vec{X}_o^{(K+1)}$ be the equivalent of $\vec{X}$.

(14) Let $X_o^{(K+1)} = X_{n+1}^{(K)}$

FOOTNOTES

1. The author of Reference (1) used the gap at the central point on the circumferential oil sealing surface as the average gap. This basis is somewhat incorrect, especially when the eccentricity ε is large, the error is large.

   The author of Reference (3) corrected this and used the average value of the gaps on the circumferential oil sealing surface as the average gap, but a mistake was made in the derivation. The main mistakes were taking

   $$\overline{\int_{\theta_n-\frac{\beta}{2}}^{\theta_n+\frac{\beta}{2}} H\,d\theta}^{\,-1} \quad \text{as} \quad \int_{\theta_n-\frac{\beta}{2}}^{\theta_n+\frac{\beta}{2}} \frac{d\theta}{H};$$

   and taking

   $$\left[\frac{1}{\beta}\int_{\theta_n-\frac{\beta}{2}}^{\theta_n+\frac{\beta}{2}} c(1+\varepsilon\cos\theta)\,d\theta\right]^3 \quad \text{as} \quad \frac{1}{\beta}\int_{\theta_n-\frac{\beta}{2}}^{\theta_n+\frac{\beta}{2}} c^3(1+\varepsilon\cos\theta)^3\,d\theta$$

2. It is not difficult to derive $\bar{p}_n = \frac{1}{(1+\xi)}, \xi = \frac{R_i}{R_o}$.
   R_i ----- resistance coefficient of the throttle, R_o ----- resistance coefficient of the oil sealing surface, thus for $\bar{p}_n$ here, when compared to the resistance in Reference (5), ξ has the same physical meaning.

3. If the friction resistance caused by the pressure gradient in the oil recess is considered, see Reference (4).

4. Figure 6 - Figure 9 are taken from Reference (1).

5. Figures 10-15 are taken from Reference (3).

6. Stiffness J is defined as $J = \left|\frac{\partial W}{\partial e}\right|$ or $\left|\frac{\partial \bar{W}}{\partial \varepsilon}\right|$

   since the optimum design here has taken ε as a predetermined parameter and is given beforehand, therefore J_{max} and W_{max} are consistent.

7. For definite dimensions of the oil recess, the nondimensional load capacity $\bar{W}$ is the monotonic continuous function of the design parameter λ (or δ) of the throttle (see Figure 6, Figure 7), thus based on

   $\frac{\partial \bar{W}}{\partial \lambda\,(\text{或}\,\partial \delta)} = 0$, the λ (or δ) of maximum load capacity can be obtained.

8. Although the author of Reference (3) used Powell's method, it was used only to adjust one variable. This can actually be realized by a one-dimensional search, thus Reference (3) used Powell's method in form, but actually it was only a method of rotation of coordinates.

9. According to Reference (10), the two complex conditions here can be equivalently substituted by the following singular simple condition

$$\left(\tfrac{1}{2}[(f(X_{n}) - 2f(X_{m}) + f(X_{m+1})] > \Delta f \right) .$$

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TITLE: "On the Invariant Set and Periodicity of the Solution of a Class of Probability Measure Equations"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 1-11

TEXT OF ENGLISH ABSTRACT: Let the phase space of a stochastic process be a metric measurable space. At time t, denote the probability distribution of a stochastic process with the initial distribution ϕ by $m(t, \phi, \cdot)$. We assume that the family of probability measures $\{m(t, \phi, \cdot), t \geqslant 0\}$ satisfies the semi-group property. Consider the equation of the one-parameter measure family:

$$\psi(t + s, \cdot) = m(s, \psi(t), \cdot), \ t \geqq 0, \ s \geqq 0.$$

$$\psi(0, \cdot) = \psi_0(\cdot).$$

There exists a unique family of solution measures for this equation. In this paper we discuss some properties of this measure family.

In the second section the invariance theorem is proved after several lemmas are established. On the time interval $[0, \infty)$, we prove that the limit set of

[Continuation of NANJING DAXUE XUEBAO No 4, 1979 pp 1-11]

$\{m(t, \phi, \cdot), t \geqslant 0\}$ is a non-empty, weakly bounded and weakly compact invariant measure set under some conditions. The proof of this theorem is simpler and the conditions weaker than those given by Kushner.

In the third section we consider a time-homogeneous Markov process associated with a stochastic continuous transition function. We prove that there are only three possible cases for the solution of the measure equation: the equilibrium measure or the family of periodic measures from certain moment on, or the measures of the solution are different from each other.

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TITLE: "On the Design and Implementation of the Systems Programming Language XCY"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 13-21

TEXT OF ENGLISH ABSTRACT: The systems programming language XCY is chiefly designed for writing systems programs, e.g., compilers and operating systems. The production rate of computer software may be prominently increased by using this language as a production tool. In this paper are emphasized some important features of its design and implementation, especially its design objective and requirements, data structures, modules, paths, and the storage scheme used in the DJS-200 XCY compiler.

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TITLE: "Conserved Currents and Weak Neutral Current Interactions"

SOURCE: Nanjing NAN[illegible] XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 23-30

TEXT OF ENGLISH ABSTRACT: Assuming that the conserved currents are sources of the vector meson fields, the vector fields Z's coupled to baryon and lepton number are discussed. The linear combinations of baryonic, leptonic and electromagnetic currents are related to weak neutral currents, which fit all the present neutral current experiments. Several theorems concerning e-μ-τ asymmetry of Z meson's couplings have been proved; the e-μ-τ mass differences are thus accounted for by a corresponding mechanism.

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TITLE: "The Investigation of the Physical Condition of Prominences"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 31-36

TEXT OF ENGLISH ABSTRACT: In this paper we introduce macroscopical motion of mass and prove convincingly the conclusion by Dubov that the turbulence velocity V_t is correlated with μ. Therefore, it is reasonable to determine the temperature of the prominence with hydrogen and helium lines. Using the data of Ye, we calculate and eventually tell the physical features of the hydrogen prominence from the metal one. For the former, $T_e = 7340$, $V_t = 4.3$, $N_2 = 2 \times 10^{13}$; for the latter, $T_e = 5520$, $V_t = 6.7$, $N_2 = 1 \times 10^{14}$. It is found that the prominence at higher temperatures corresponds to lower density, and vice versa, like chromosphere.

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TITLE: "A New Approach to the Representation Theory of Finite Groups (III)--Application of Eigenfunction Method to Point Groups"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 37-48

TEXT OF ENGLISH ABSTRACT: The "Eigenfunction Method" is used to treat the molecular symmetries; a simple method is given for constructing Symmetry-Adapted-Linear-Combinations in the single electron case. The so-called "intrinsic quantum numbers" are introduced to serve as the missing labels in multiplicity cases.

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TITLE: "Studies on Alkali Metal Promoted Iron Catalyst of Ammonia Synthesis"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 49-58

TEXT OF ENGLISH ABSTRACT: When iron oxalate carried by active carbon was first reduced and then plated with metallic potassium, the ammonia synthesis catalyst thus formed exhibited especially high activity. After an intensive study of each component of the catalytic system, the following results have been obtained:

Mossbauer spectrum revealed that the active site of the catalyst was composed of particulate α-iron having sizes around 100 Å. The active center was in the (111) plane of α-iron with seven atoms forming a Fe_7-cluster in which activation of dinitrogen took place. EHMO method of calculation showed that Fe_7-cluster was the most efficient structure of iron catalyst to activate dinitrogen and it reduced the bond overlap population from 1.640 to 1.004, corresponding roughly to the reduction of the triple bond to a double bond. It was the π-bond that was greatly

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weakened while the σ-bond remained nearly the same.

Metallic potassium in the catalyst formed potassium hydride with dihydrogen, first having activated dihydrogen. Results of the SCCC method of calculation of FeH showed that H in the hydride bore a charge of 0.199 electron. Action of active carbon, the carrier, was independent of the degree of its graphitization, but dependent on its specific surface and distribution of its pore size. A pore size around 100 Å was most suitable for the formation of the Fe_7-cluster. As for the other kinds of carriers, either because of having too small a bore size (e.g., molecular sieve) or because of the oxygen constituent (e.g., SiO_2, Al_2O_3, etc.), they were unfavorable for the formation or function of the Fe_7-cluster. Chloride-ion was a poison for the iron catalyst of ammonia synthesis because it caused the agglomeration of the superfine particles of α-iron and hence destroyed the active site with the result of the loss of activity.

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TITLE: "The Mossbauer Spectroscopy Research of Fe-Cr Catalyst under the Reaction Condition"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 59-64

TEXT OF ENGLISH ABSTRACT: Through the measurements of Mossbauer spectra under the reaction condition (350°C, water:gas = 2:1, velocity of gas = 250 ml/hr), it is shown that S_8 belongs to the $Fe_{3-x}Cr_xO_4$ system. After the treatment at 550°C the area of the paramagnetic spectrum increases. It indicates that a more fine-particle solid solution is formed. Raising the space velocity the reduction is incomplete and γ-Fe_2O_3 is formed. At this condition γ-Fe_2O_3 may be the intermediate phase of α-Fe_2O_3 reducing to Fe_3O_4. The increasing in amount of fine-particle solid solution results in the rising of the catalytic activity.

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TITLE: "The Chemical Kinetics of Direct Hydrolysis of N-Carbo-Benzoxy-L-Tyrosine P-Nitrophenyl Ester by Two Molecular Forms of Urokinase"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 65-73

TEXT OF ENGLISH ABSTRACT: When using plasminogen as the substrate, the Km value for 54,000 mol. wt. urokinase was half the value for the 33,000 mol. wt. urokinase. When using CTN as the substrate, the Km value for the 54,000 mol. wt. urokinase and the 33,000 mol. wt. were 5×10^{-4} M and 4×10^{-4} M respectively.

The effect of pH on the enzyme-catalyzed hydrolysis of CTN ester has been found to be of the non-competitive type for both the high molecular urokinase and the low molecular urokinase. This fact showed that in the present case, $k_{-1} \gg k_2$ or $Km = \frac{k_{-1}}{K_1}$, i.e., the Km was actually the reciprocal of the equilibrium constant for the formation of the enzyme-substrate complex. The pK value of the ionizable group of the active center was evaluated by LogV-pH plot. The high molecular urokinase had the pK value of 7.65 and that of the low molecular urokinase was 7.85.

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The E-aminocaproic acid acted as a competitive inhibitor of both high molecular urokinase and low molecular urokinase. The evaluated dissociation constant Ki for the assumed H-UK-EACA complex was 0.8×10^{-2} M and 1.5×10^{-2} M for the L-UK-EACA complex. The CTN hydrolysis activity of the forms of urokinase was not affected by natural inhibitors of trypsin of different sources.

The influence of temperature on both the Km and V for the hydrolysis of CTN has been determined and the thermodynamic constants for the equilibrium reaction have been evaluated.

The kinetic and thermodynamic data indicated that the active center of the two forms of urokinase was identical in chemical nature, but their conformations differed from each other. The conformation of the high molecular urokinase facilitated the catalysis of CTN hydrolysis with higher efficiency.

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TITLE: "The Texture and Structure of Ores from the Sulfide Deposits of Qixiashan, Nanjing, and Their Implication on the Ore Genesis"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 75-94

TEXT OF ENGLISH ABSTRACT: The pyritic lead-zinc deposits of Qixiashan in the Nanjing Hills, some 20 km east of Nanjing in Jiangsu Province, discovered in 1950, have long been thought to be undoubtedly epigenetic and hydrothermal in origin. The writers recently studied the deposits in detail. There were much interested when they found different sorts of framboidal pyrite in the concretionary and also in the laminated ores of the deposits. Some of the framboidal pyrites have the distinct cellular structures of fossilized tissues of algae, and some can be identified to be *Solenapora*. The framboids are now made up mainly of pyrite and melnikovite. The interstices between the "cells" are mostly filled with organic matter. The internal structures of the cellular tissues are of two kinds: (1) the cells, about 20μ in size, are arranged in radiating siphon-like bodies, and (2) the cells, of 40 to 60μ in size, are arranged orderly. Associated with the iron

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sulfide framboids, there are also some metacolloidal structures composed of marcasite and wurtzite. It is probable that the sulfides were formed when the decaying biogenic protoplasm occurred in the presence of sulfur-reducing bacteria to produce HS^-, which combined with Fe^{++} to yield framboidal melnikovite and its recrystallized product, pyrite. Therefore, they were formed by biochemical processes as a result of syngenetic sedimentation and early-diagenetic products.

The ore bodies are mostly stratiform, stratoid, or in large lenticular forms. The ores are laminated, banded, or in concretionary layers. These structures of the ore bodies are mainly parallel to the stratification of the country rocks. Such kinds of ore constitute the main industrial ores of this district. The age of the ore formation belongs to the early to middle Carboniferous period.

But there are also evidently hydrothermal effects which are superimposed upon the syngenetic ore-forming processes, resulting in the formation.

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TITLE: "Karst of China and Principles of Its Development"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 95-108

TEXT OF ENGLISH ABSTRACT: In China, carbonate rocks cover an area of more than 1.3 million km^2, or about one-seventh of the total area of the country. China ranks among the most typical and extensive karst countries of the world.

(1) Types of Karst
Karst of China may be grouped into three main types, i.e., tropical, subtropical (or Mediterranean type) and temperate.

Tropical karst is dominated by mogotes-cockpits, which form characteristic landscape of southern Guizhou and northwest Guangxi. Karst of the temperate zone is characterized by dry valleys and large karst springs, with well developed karst fissures underground, but few surface karst forms. Subtropical karst consists of gentle karst hills and depressions.

The great difference between karst of different climatic zones is mainly due to bio-climatic factors. It is well known that biogenic CO_2 is important in the development of karts. In the tropical rainforest of south China, the amount of fallen leaves and branches per year is about three times that of the temperate mixed forest. Therefore, the CO_2 content of soil air in laterite is about two to five times higher than that of red soils (central China) and black soils (north China). As a result, the rate of corrosion in central Guangxi is two to five times higher than that of the Changjiang basin, and 7 to 10 times higher than that of north China. Evidently, the development of typical tropical karst forms, such as mogotes, cockpits and "stone forest" (or crevice karst), owes much to this cause.

(2) Paleo-karst

In China, the distribution of tropical karst far exceeds the limit of the present tropical zone. Tropical mogotes are widely distributed in southern Hunan, southern Guizhou and eastern Yunnan where the present climate is subtropical. In the famous "stone forest" region of Lunan, depressions between limestone pinnacles are filled with sediments of old Tertiary age. This fact shows conclusively that these tropical "fossil" karsts must have developed in early Tertiary, but under the present climatic conditions they have been gradually modified and reduced to less than 100 m high. They are known as relict mogotes.

The discovery of old tropical mogotes in the Tibetan Plateau is one of the most important contributions of karst research in China. They are widely distributed

in the Tibetan Plateau, reaching an altitude of 5200 m above sea level, the highest mogote known in the world. There were also formed in Tertiary when the climate of Tibet was moist and hot, as proved by Miocene-eary Pliocene Hipparion-Samotherium sp and paleotragus microdon fauna recently discovered in Tibet. But under the present periglacial climatic regime, these Tertiary mogotes have been mostly destroyed by frost action and solifluction and their ended remnants stand only as isolated steep limestone hills about 20-30 m high. It is not an accidental phenomenon that these old mogotes have been discovered widely in the elevated plateaus north of the Himalaya-Alpine mountain system in the world, i.e., from the Tibetan plateau westward through Pamir, Caucasus, southern Poland to the Swabian Alps of Germany, all situated along the coast of the Tethys Sea and influenced by the tropical Tethys current. This fact agrees well with the main theme of plate tectonics.

(3) The Influence of Rock, Structure Topography

1) Carbonate rocks In addition to chemical composition, rock texture also exerts an important influence on karst. For example, in central Shandong, karst is better developed in some dolomite and clayey-dolomitic limestone karst is better than pure limestone because the former consists of bioclastic limestone oolitic limestone and grainstone. In central Guizhou and Guangxi, karst is usually better developed in bioclastic limestones.

2) Structure It is well known that structure has a distinct influence on the development of karst. A remarkable example is karst of eastern Sichuan where,

superposed on the general pattern of relief of parallel ridges and valleys due to close folding of the northeast-southwest direction, are narrow karstic "trough valleys" on the top of anticlinal ridges. These "trough valleys," up to more than 100 km long, developed in the axis of anticlines due to the outcrops of limestones there. This particular type of karst morphology is known as Eastern Sichuan karst in China.

3) Topography The long-established principle that the degree of karstification is more pronounced in river valleys than on water divides does not agree with actual facts in China. In the Yunnan-Guizhou plateau and mountains of the border of Sichuan, Hunan and Hubei, it is generally found that karst is more strongly developed in water divide areas. This is probably due to the fact that these water divides belong to karst peneplains of old Tertiary age (or Pre-Tertiary) and karstification has continued from that time to the present, so that recent karst is superposed on old karst, resulting in more pronounced karst features both on the surface and underground.

The correlation between caves and river terraces should be used with great caution. There are a number of caves in China where the altitude of caves does not correspond to that of river terraces. The reasons for this are varied and are enumerated in the Chinese text.

(4) Deep Karst

During the last two decades, a large number of deep caves, situated far below sea

level or the land surface, has been found in China. They may be grouped under four categories:

1) Deep caves within recently uplifted plateaus or mountains in Guizhou, found at 400 m below the summit surface, whereas in northern Sichuan they lie at 800 m below the summit surface. They are mostly vertical caves filled with clay or mud, probably corresponding to the clay filling stage of cavern development in an uplifted peneplain.

2) Deep caves close to sulfide ore bodies in pyrite, galena and copper ore regions of China, of which a line of deep caves is often found in the contact zone between carbonate rock and ore bodies. One deep cave in Anhui lies at -350 m. Evidently this is chiefly due to the fact that during the oxidation of sulfide ores, a large amount of SO_4 ions is formed which greatly increases the corrosive power of the ground water.

3) Deep caves formed by pressure water with a steep hydraulic gradient or siphon circulation, such as caves at 10 or more than 100 m below the river bed in the gorges of the Changjiang and Wujiang.

4) Buried caves of ancient geological age. In a coal mine of central Guangxi, deep caves at -700 m were probably formed in the Permian. However, as they are located near the axis of a syncline, they still act as conduits of present ground water circulation. At present, these deep caves are largely void; however, some are filled with silt and silty sand which is not cemented and has distinct micro-cross bedding. This indicates that the ground water at such great depth is still a turbulent flow. Therefore, although products of the geological past, these deep

[Continuation of NANJING DAXUE XUEBAO No 4, 1979 pp 95-108]

caves are continuously being reshaped by the present ground water circulation. Deep caves below the north China plain are probably of the same character.

The origin of deep caves is attributed to the fact that ground water at great depth still possesses considerable corrosive power due to (1) relatively high velocity of deep ground water in favorable structure setting, (2) its high content of CO_2, (3) its comparatively high content of SO_4 ions and (4) the effect of mixing corrosion. The above factors are all substantiated by the Chinese data of veins, veinlets, networks and some drussy ores. The veins have the characteristics of fracture-filling and metasomatism accompanied by silicification and baritization. In such ores, commonly the pyrite is partly replaced by sphalerite which, in turn, is partly replaced by galena. Thus, the lead-zinc ores of this district are mainly related to this mode of mineralization. Taken as a whole, the genesis of the ores of this district should be considered as syngenetic sedimentation superimposed by hydrothermal effects.

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TITLE: "The Variation of Flood and Drought in the Latest Centuries in Eastern China"

SOURCE: Nanjing NANJING DAXUE XUEBAO [JOURNAL OF NANJING UNIVERSITY], Natural Sciences Edition in Chinese No 4, 1979 pp 109-121

TEXT OF ENGLISH ABSTRACT: The data analyzed in this paper were produced from the two historical climatic records seminars which were held at the Geophysical Department of Beijing University and the Meteorological Department of Nanjing University respectively in 1974 and 1977 under the auspices of the Research Institute of the Central Meteorological Bureau (now called the Academy of Meteorological Science of the Central Meteorological Bureau). The paper mainly analyzes the following topics: 1. Variability of flood and drought; 2. Persistence and interchangeability of flood and drought; 3. Periods and cyclic periods of drought and flood; 4. Regional variation and types of flood and drought; 5. The variation and trend of flood and drought.

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TITLE: "Multiplex Photogrammetry - A Novel Reconnoitre"

SOURCE: Beijing HAIYANG [OCEANS] No 9, 15 Sep 80 pp 25-26

ABSTRACT: Based on a recollection of the allied landing on Okinawa in 1942 of the Second World War, the authors introduce the principles and performance theory of multiplex photogrammetry, where as much as nine filter lens covering a wide range of wavelengths are incorporated into survey cameras, which distinguishes all types of objects and variations at various depths of coastal sea water, and provides clear pictures of ocean floor topography , hidden reefs, and water depths, besides detection of petroleum pollution in oceans. It is reported that, in barely three years, China has successfully self-developed two models of such cameras that cover the ranges of ultraviolet, visible and infrared.

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TITLE: "Prof. Yu's Conversation on Space Survey Ships"

SOURCE: Beijing HAIYANG [OCEANS] No 9, 15 Sep 80 pp 29-30

ABSTRACT: The author reports a visit to a certain ship design and planning laboratory, where Prof. Yu explained and illustrated with photographic slides, models and tape recordings the functions of such space survey ships as target survey ships, satellite tracking ships and space survey ships in the monitoring, remote control and communication of intercontinental missiles and space crafts. The characteristics of radar antenna and ship stabilizingequipment, as well as the design of various facilities indispensible to such type of ships are also described.

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